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(54) Title: ENGINE

(57) Abstract

A constant pressure Brayton cycle engine (40) including a combustion chamber (46) and a positive displacement compressor (42) connected by a shaft (41) to an output wheel (44). Gas is fed to the compressor (42), then to the combustion chamber (46) and then to the output wheel (44). The engine (40) can be either open cycle or closed cycle. The compressor (42) can be of any suitable type such as a sliding vane, a Roots type blower (sometimes called a screw or gear compressor), a regenerative blower, or a liquid seal (Nash) blower. The output wheel (44) can be any of these types or also a dynamic (axial or centrifugal) turbine wheel.

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ENGINE

CROSS-REFERENCE TO RELATED APPLICATIONS

This application is a continuation-in-part
of applicant's copending application serial No. 879,969,
filed February 21, 1978 and entitled "Gas Turbine System";
and is also a continuation-in-part of applicant's
copending application serial No. 890,465, filed March 27,
1978 and entitled "Gas Turbine System". Each of these
two parent applications in their entirety is hereby
incorporated by reference in this application.

TECHNICAL FIELD

This invention relates to engines or prime
movers and in a preferred embodiment to a constant
pressure Brayton cycle engine using a positive dis-
placement compressor.

BACKGROUND OF THE PRIOR ART

All previous methods of adapting the Brayton
(or Joule or constant pressure) continuous cycle for
prime mover service have centered around the classical gas
turbine concept. A gas turbine includes three devices:
A dynamic (non-positive displacement) compressor of the
axial or centrifugal type; a fuel combustor of the direct
of indirect type; an output turbine (power wheel) of the
impulse or reaction type. The gas turbine approach has
many advantages at high power levels (central station
power generation, aircraft engines, etc.), but suffers
serious economic handicaps at the lower horsepower levels
(up to about 300 h.p.) required by automobile service.

The compressor and turbine share a common shaft
in the classical gas turbine engine. Thus, the turbine
(in a single shaft engine) must drive its own compressor
in additional to driving the load. The reason for this is
that only a gas turbine is capable of the ultra-high r.p.m.
required by the previously used, high speed, dynamic com-
pressors. A simple single shaft, classical engine is
shown in Fig. 1

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and comprises a compressor 12, a turbine 14, a shaft 16
connecting the compressor and turbine and a combustion
chamber 18. Fig. 2 shows two cases of different compressor-
to-output power ratios (Pc/Po). Fig. 2A shows a compressor

5 20 and a turbine 22 connected together by a shaft 24. The
arrangement shown in Fig. 2A has a compressor-to-output
power ratio of 0.5. Thus, a 100 h.p. output requires
a 0.5 x 100 = 50 h.p. compressor. It follows that the
power turbine for this case must supply 50 + 100 = 150 h.p.

10 Fig. 2B shows a compressor 26 and a turbine 28 connected
together by a shaft 33. In Fig. 2B the compressor-to-output
power ratio is 2.0. The 100 h.p. output power for this
case requires a 200 h.p. compressor and a 100 + 200 = 300 h.
p. turbine. Clearly the overall equipment size, for a given

15 output shaft power, depends upon the required compressor
size. It can be shown (see "Aircraft Gas Turbines" C. W.
Smith John Wiley - page 43 or "Propulsion Systems" A. N.
Hosny, University of South Carolina Press - page 51)
that the compressor to output power ratio can be simply

20 stated as:

$$1. \quad \frac{Pc}{Po} = \frac{1}{\eta_T \, \eta_C \, \frac{1}{x} \left(\frac{T_3}{T_1}\right) - 1}$$

where:

Pc/Po = Compressor To Output Horsepower Ratio

25 T_3/T_1 = Combustion To Inlet Temperature Ratio
(°F, Absolute)

η_T = Thermodynamic Efficiency of Turbine

η_C = Thermodynamic Efficiency of Compressor

x = $\left(\frac{P_2}{P_1}\right)^{\frac{\partial - 1}{\partial}}$

30

P_2/P_1 = Compressor Outlet To Inlet Pressure
Ratio

∂ = Ratio of Heat Capacity At Constant
Pressure To That At Constant Volume

For a fixed ambient temperature (T_1) equation 1 states

35 that the compressor-to-output power ratio depends only
upon the pressure ratio adopted and the combustion tempera-
ture, since the other parameters, η_T, η_C, and ∂ are essentially
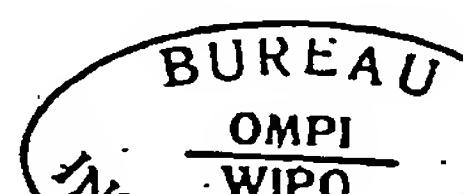
fixed.   Equation 1 is plotted in Fig. 3 and provides a
good overall picture of the classical problem.   Anything
above a compressor-to-output power ratio of 1.0 means
a very large compressor indeed.   The parameters from Fig. 3
5 can be transformed to practical values in order to illustrate
the classic problem involved and the solution provided by
this invention.   Consider the case of 50 h.p. net output.

The data from Fig. 3 can now be translated to absolute com-
pressor ratings and this is plotted in Fig. 4   Typical
10 materials in today's technology allow for a tempera-
ture ratio (T_3/T_1) of about 3.   Thus, Fig. 4
shows a marked sensitivity to pressure ratio.
Absolute efficiency considerations dictate a pressure ratio
in the range of 4:1 to 6:1.   If a temperature ratio of
15 6 could be tolerated, Fig. 4 shows an insensitivity of
compressor requirements to pressure ratio.   If we assume
an ambient temperature of 70°F, then a clearer picture of
the problem emerges and is shown in Fig. 5.   The steepness
of the curve in the vicinity of 1000°F is the cause of
20 the major problems as will become clear from the next curve.
A careful review of commercially available
compressors of different types allows the compressor weight
vs. horsepower curves of Fig. 6 to be drawn. The high r.p.m.
associated with classical gas turbines limits severely
25 the tolerable turbine inlet temperature.   The relatively
cheap materials that must be used for automobile service
sets a limit of 1000-1500°F.   Fig. 5 dictates a compressor
in the 50-100 h.p. class to supply a net output of 50 h.p.
Consider now Fig. 6:   A piston compressor is completely
30 out of the question since it would weigh between 1500 and
2000 pounds.   The Roots type blower and sliding vane
type compressors are better but also are in the impractical
range of 200-600 pounds to supply 50 h.p. of shaft output
power.   Only the non-positive displacement compressor has
35 a reasonable weight for the classical approach.   It is
this compressor weight factor alone - not cost nor effi-
ciency but simply compressor weight that forces classical gas
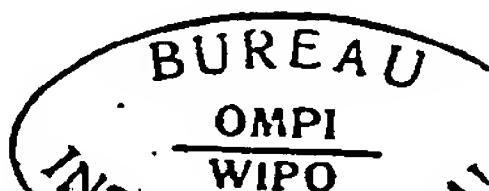
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turbines to use the non-positive displacement compressor
to supply the massive amounts of compressed air that are
required.

The penalty paid for achieving the high
5 compressor h.p. to weight ratio for the classic gas turbine
is severe indeed:

1. The ultra-high r.p.m. is many times greater
than the automobile r.p.m.

2. The ultra-high r.p.m. of the hot turbine
10 gives rise to excessive tensile stress
due to centrifugal forces. This limits
both usable materials and allowable inlet
temperatures.

3. The compressor mass flow and output
15 pressure vary with r.p.m. The fact that
pressure varies means part-load efficiency
is poor.

It is important to note that the output turbine requirement
does not dictate a high r.p.m., per se; the high r.p.m.
20 is required strictly to satisfy the compressor needs. In
fact, there is no other method (electric motor drive, for
example) of achieving the high r.p.m. required by the
compressor.

This then is the state of affairs for gas turbines
25 in automobile service. Fuel economy dictates both a high
inlet temperature and a relatively high pressure ratio
(about 4:1). Material limitations prevent high combustion
temperatures, which in turn results in both low efficiency
and high compressor requirements (Fig. 5), thus practically
30 eliminating all but high speed, turbo-compressors (Fig. 6)
on a weight basis alone. It is important to note that
jet aircraft, central station power, and other non-auto
applications can afford to use more expensive materials and
devices and hence can utilize somewhat higher combustion
35 temperatures. A jet aircraft turbine may cost $50,000 to

build, but an automobile gas turbine must be limited
to the $100-$200 class. As a result, gas turbines have
found widespread service outside the automobile field
only.

5   The main thrust of gas turbine technology
today is towards better and cheaper high temperature
materials and assemblies, super-alloys, exotic cooling
methods, ceramic turbine buckets, cermet technology,
plasma coatings, etc.

10   SUMMARY OF THE INVENTION

           The present invention is a Brayton cycle
engine method and apparatus using a positive displacement
compressor. The Brayton cycle engine of the present
invention includes a positive displacement compressor,
15   an output wheel connected by a shaft to the compressor,
a combustion chamber, and means for feeding a gas to the
compressor, means for feeding compressed gas from the
compressor to the combustion chamber, means for burning
20   fuel in the combustion chamber, means for feeding hot
gas from the combustion chamber to the output wheel,
and means for feeding exhaust gas from the output wheel.
The compressor can be any suitable positive displacement
compressor such as a sliding vane compressor, a Roots
25   type blower (sometimes called a screw or gear compressor),
a regenerative blower, or a liquid seal (Nash) type blower.
The output wheel can be any one of the above mentioned
devices or in addition can be a dynamic turbine wheel.
           This removes the classic problems associated
30   with high speed, non-positive displacement compressors:
high cost and complexity, high stress on hot turbine
buckets, low efficiency at part-load, etc. Use of a
low speed, positive displacement compressor is made
possible by the use of high combustion temperatures as
shown in Fig. 5. These high combustion temperatures are
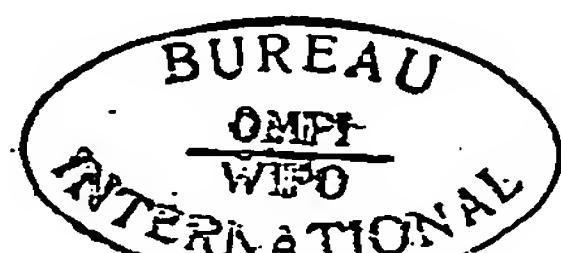
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basically made possible by either one of two methods:

1. Use of an intermittent burn-cool operating cycle as described in copending application serial No. 879,969; and

2. Use of a bootstrap argument. This results from the basic understanding provided by Figs. 1 through 6. The adoption of a lower r.p.m. means a lower tensile stress on the hot turbine bucket. The reduced stress permits an increased temperature since high temperature creep is a main limitation. The increased temperature results in reduced compressor requirements (Fig. 5) which in turn makes positive displacement compressors (low r.p.m.) feasible. The net result is a cheaper, more efficient and flexible implementation of the Brayton cycle.

Of course, as new high temperature materials, devices, and techniques became available they can be incorporated into the present inventions. However, it is to be clearly understood that the present invention can be practiced with today's materials, devices, and technologies.

It is desirable to expand Fig. 6 to the 0-50 h.p. range of interest to automobile service. This results in Fig. 7. However, since Roots blowers, sliding vane compressors, helical screw compressors, etc. have not been engineered with the present problem in mind, it is felt that improvements can be readily made to transform the data in Fig. 7 into the curves of Fig. 8. The basic idea to be conveyed is that virtually any type of positive displacement compressor can be used to implement the present invention.

It is an object of the present invention to overcome the disadvantages of the Brayton cycle gas turbine

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by providing a non-gas turbine implementation of the
constant pressure Brayton cycle. It is another object
of the invention to provide a Brayton cycle engine having
a positive displacement compressor.

5   BRIEF DESCRIPTION OF THE DRAWINGS

        The present invention will be more fully
understood by reference to the following detailed descrip-
tion thereof, when read in conjunction with the attached
drawings, wherein like reference numerals refer to like
10 elements and wherein;
            Fig. 1 is a partly diagramatic, partly schematic
view of a prior art Brayton cycle gas turbine;
            Figs. 2A and 2B are diagramatic views of a
compressor-turbine-shaft combination;
15   Fig. 3 is a graph of the ratio of compressor
power to output shaft power vs. pressure ratio.
            Fig. 4 is a graph of required compressor power
(h.p.) vs. pressure ratio;
            Fig. 5 is a graph of required compressor power
20 (h.p.) vs. combustion temperature (^{O}F);
            Fig. 6 is a graph of compressor weight (lbs.)
which is compressor power (h.p.);
            Fig. 7 is a graph of compressor weight (lbs.)
vs. compressor power (h.p.);
25   Fig. 8 is a graph of compressor weight (lbs.)
vs. compressor power (h.p.);
            Fig. 9 is a partly diagramatic, partly schematic
view of an engine according to one embodiment of the present
invention including a compressor, an output power wheel
30 and a combustion chamber;
            Fig. 10 is a partly diagramatic, partly schematic,
view of an engine according to another embodiment of the
present invention including a compressor, an output power
wheel and an indirect combustion chamber for use in a
35 closed cycle operation;
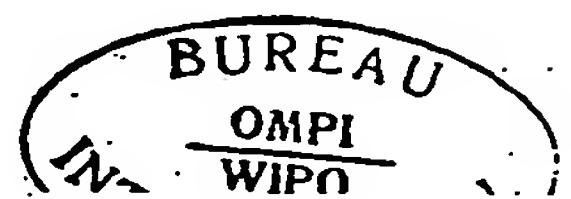
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Fig. 11 is a graph showing desirability vs. horsepower, speed, or torque;

Fig. 12 is a graph showing tensile strength vs. temperature for selected graphites;

5   Fig. 13 is a graph showing vaporization rate vs. temperature for graphite;

Fig. 14 is a graph showing pressure and flow rate vs. r.p.m. for a dynamic turbo-compressor; and

Fig. 15 is a graph showing pressure and flow

10   rate vs. r.p.m. for a positive displacement compressor.

DETAILED DESCRIPTION OF THE PREFERRED EMBODIMENTS

With reference to the drawings, Fig. 9 shows a substantially constant pressure Brayton cycle engine 40 according to the present invention comprising a

15   positive displacement compressor 42 and an output power wheel 44 connected to the compressor by a shaft 41 and also connected to an output shaft 48. The engine 40 includes a combustion chamber 46 to which fuel is fed by a fuel line 47. Air is fed into the compressor 42 through

20   an air inlet 43 and compressed air is fed from the compressor 42 to the combustion chamber 46 through an air line 49. Compressed, hot gas is fed from the combustion chamber 46 through a continuation of the gas line 49 to the output power wheel 44 from which the exhaust gas is

25   fed to ambient through an exhaust line 45. The engine 40 is an open cycle engine as will be clearly understood by those skilled in the art.

Fig. 10 shows a closed cycle engine 50 according to another embodiment of the present invention. The engine

30   50 includes a positive displacement compressor 52 and an output power wheel 54 connected to the compressor by a shaft 56. The output power wheel 54 is also connected to an output shaft 60. The engine 50 also includes an indirect combustion chamber 58 into which air and fuel

35   are fed by lines 68 and 70 respectively, and from which

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the exhaust gas is fed to ambient through an exhaust
line 72. Compressed gas from the compressor 54 is fed
by a line 62 from the compressor to a heat exchanger
74 (such as a coil) in the indirect combustion chamber

5 58 and from there to the output power wheel 54. The
exhaust from the output wheel 54 is fed by a line 64
back to the compressor 52. The exhaust gas from the
output power wheel 54 is heat exchanged by a heat ex-
changer 66 with the gas fed from the compressor 52 to the

10 combustion chamber 58. The various advantages of the
closed cycle are described in the parent applications
incorporated herein by reference above. It is to be noted
that in the open cycle embodiment of Fig. 9, the hot
gas fed to the output power wheel is the products of

15 combustion of the burning fuel in air. In the closed
cycle embodiment of Fig. 10, on the other hand, the hot
gas fed to the output power wheel is whatever is chosen
for the working gas, such as nitrogen, neon, carbon
dioxide, etc.

20   According to the present invention, the engines
40 in Fig. 9 and 50 in Fig. 10 can use for the compressor
any positive displacement compressor such as, for example:
(1) a sliding vane compressor, (2) a Roots type blower
(sometimes called a screw or gear compressor), (3) a

25 regenerative blower, or (4) a liquid seal (Nash) type
blower.

          The output power wheel in each embodiment
can also use any one of the above listed types of devices
and in addition can use a dynamic turbine wheel. Thus,

30 20 different combinations are preferred and each has its
own special application. For example:

          1. Rotary vane/Rotary vane - is preferred
              for small car automotive service;

          2. Rotary vane/Roots blower - is preferred

35   for truck service;

3. Roots blower/Roots blower - is preferred for low speed, high torque service (farm tractor, etc.)

In the above, the first name refers to the compressor and the second to the output power wheel or motor. Thus, a rotary vane/roots blower configuration means a rotary vane compressor and a roots blower run backwards as a motor. It should be noted that compressors run backwards act as efficient motors, see, for example, "Pneumatics and Hydraulics" by H. L. Steward (Audel and Co.) illustrate this point. The Gast Manufacturing offers sliding vane motors for sale in the 1-10 h.p. class (Models 16AM-FCC-1).

These configurations result in a desirability curve vs. horsepower and/or speed and/or torque as depicted in Fig. 11.

One preferred specific embodiment is to use a sliding vane motor (output power wheel) with the vanes being made of graphite. This offers several advantages. Graphite is a good lubricant in itself and thus the wear on the vanes is minimized. Second, graphite is known to increase in tensile strength as temperature is increased to about 4500°F as shown in Fig. 12. Thus, this feature allows combustion temperatures up to about 4500°F to be used in the expansion output wheel. Fig. 12 is a graph of ultimate tensile strength vs. temperature for selected graphites. All specimens were tested in the direction of major anisotropy. (1) Petroleum coke base, fine grain, extruded, d = 1.67; (2) lampblack base, molded, d = 1.50; (3) petroleum coke base, medium grain, extruded, d = 1.55; (4) petroleum coke base, fine grain, molded, d = 1.75.

Graphite is known to have a relatively high vapor pressure at high temperatures. This will result in a vaporization rate. Assuming 100,000 miles of service life, and 50 miles/hour average speed, then 7×10^6 sec.

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of service life is required. If 10% of the graphite sur-
face is allowed to vaporize during this period, then a
vaporization rate of about 10^{-8} gram/cm^2 sec. is tolerable.
From Fig. 13 (which is the free vaporization rate of
5   graphite) it is seen that an average temperature of about
$3600^{\circ}F$ ($2200^{\circ}K$) is tolerable. While this temperature
greatly exceeds present day gas turbine capabilities, it
can be extended further by coating the graphite vanes
with a suitable high temperature metal, alloy, or ceramic.
10 In this case, vaporization will occur (if at all), only
at the tiny exposed area where the vane rubs the housing.
         While the above discussion has been with
respect to using graphite for the sliding vanes in the
motor or output power wheel of the present invention,
15 it is also desirable to use graphite for the sliding
vanes in a sliding vane type compressor. Graphite has
the advantages that it wears uniformly and conforms by
the wear process to the desired shape and that it also
provides self lubrication, is relatively inexpensive
20 and is easy to machine, etc.
         Another method of causing a major reduction
in the compressor size and weight is to utilize the closed
cycle in Fig. 10. This technique does not change the
compressor horsepower requirement. This fact is evident
25 from equation 1. The compressor-to-output power ratio
depends only upon the pressure ratio and not the absolute
gas pressure. However, at higher total pressures made
possible by the closed gas system, the compressor "swept
out volume" becomes less due to the higher gas density.
30 As a result, the key item (the compressor weight) is
reduced. This allows for an extension of this invention
to higher power levels. Thus, for example, the vane/vane
desirability curve can now take the dashed position shown
in Fig. 12.

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It should be noted that a turbine does not require the same high tip speed (high r.p.m.) as does the turbo-compressor. Thus, combinations like vane/turbine and roots blower/turbine are useful configurations. While the main thrust of this invention is the automobile,

5 it is to be understood that the engine of this invention is also applicable to trucks, locomotives, central station power, remote station power generation, emergency and stand-by power generators, aircraft, etc.

10   The present invention can be used with multiple shaft arrangements, multiple stage compressors, multiple stage expansion stages (motor output), interstage cooling, re-heat, and various techniques of heat exchange. The intermittent burn-cool operation cycle of parent application

15 serial No. 879,969, the throttle control invention of parent application serial No. 890,465, the various flywheels and compressed gas surge tanks, etc. taught in said parent applications are also applicable in this invention.

One major advantage of the present invention

20 over the classical gas turbine for automotive service is in the matching of r.p.m. The wheels of an automobile rotate at a maximum of about 1500 r.p.m. Thus, complicated speed reducers are required to match a gas turbine, at 50,000 r.p.m., to an automobile. The r.p.m. of the present

25 invention can be selected to closely match that of the auto- mobile by proper selection of compressor and power wheel diameters.

Another major advantage of this invention over the prior art is the efficiency at part-load. In fact,

30 the present teachings maintain efficiency essentially down to 0 r.p.m. This statement can best be visualized with the aid of Figs. 14 and 15. The classical turbo-compressor relies upon high impeller tip speed to provide both mass flow and pressure. Thus, a reduction in r.p.m. causes

35 both mass flow rate and pressure to be reduced as shown in

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Figs. 14. The flow rate reduction at low r.p.m. is
acceptable since low r.p.m. (part-load) requires only
a portion of the air mass flow. However, efficiency
considerations insist upon a high gas pressure, in fact,
5   optimal conditions require a narrow band of gas, pressures
(pressure ratio in 4-6 range). Thus, the low r.p.m.
drops the pressure ratio and introduces the well-known
part-load inefficiency. This is not true for the positive
displacement compressor of the present invention as shown
10 in Fig. 15. The compressor outlet pressure does not
depend upon speed but only upon the geometry of the com-
pressor apparatus. The flow rate is a direct linear
function of the r.p.m. This is the ideal state of affairs
for part-load. As more fuel is pumped to the combustor,
15 the power output wheel (sliding vane, turbine,
more output power and higher r.p.m. The higher r.p.m.
provides more compressed gas, as it should, at the same
pressure and thus the efficiency, and combustion temperature
conditions do not change.
20   The following chart may be found useful:

TYPICAL COMMERCIAL UNITS

Mfg.	Type	Horsepower		Weight (pounds)	Model No.
		Room Temp.	2500° F*		
Gast	Rotary vane	4.0	20.0	21.0	6AM-FRV-23A
Gast	Rotary vane	10.0	50.0	80.0	16AM-FRV-13
Roots (Dressler)	Roots blower	40.0	200.0	600.0	RAI-88

*Estimated, as motor
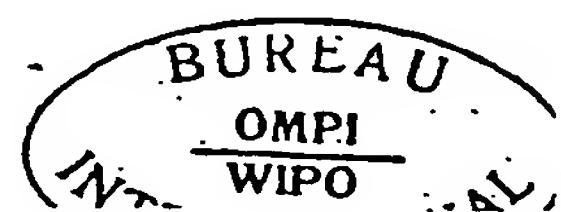
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The invention has been described in detail
with particular reference to the preferred embodiments
thereof, but it will be understood that variations
and modifications can be effected within the spirit
5   and scope of the invention as described hereinabove
and as defined in the appended claims.

What is claimed is:

1. A Brayton cycle engine comprising:

    (a) a positive displacement compressor having a gas inlet and a gas outlet;

    (b) a combustion chamber including an inlet and an outlet and means for burning a fuel therein;

    (c) an output wheel having a gas inlet and a gas outlet and being connected to said compressor by a shaft;

    (d) means for feeding a gas to said compressor inlet, means for feeding compressed gas from said compressor outlet to said combustion chamber inlet; means for feeding hot com-pressed gas from said combustion chamber outlet to said output wheel inlet, and means for feeding exhaust gas out of said output wheel outlet; and

    (e) means for operating said engine in a substantially constant pressure brayton cycle.

2. The apparatus according to claim 1 wherein said operating means comprises means for operating said engine in an open cycle and wherein said exhaust gas feeding means, feeds said exhaust gas to ambient.

3. The apparatus according to claim 1 wherein said operating means comprises means for operating said engine in a closed cycle and wherein said exhaust gas feeding means includes means for feeding said exhaust gas back to said compressor inlet, and wherein said combustion chamber is an indirect combustion chamber.

4. The apparatus according to claim 3 including means for heat exchanging the exhaust gas with the gas fed from said compressor to said combustion chamber.

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5.   The apparatus according to claim 1 wherein said compressor is a sliding vane compressor.

6.   The apparatus according to claim 1 wherein said compressor is a gear type compressor.

7.   The apparatus according to claim 1 wherein said compressor is a regenerative blower.

8.   The apparatus according to claim 1 wherein said compressor is a liquid seal blower.

9.   The apparatus according to any one of claims 5, 6, 7 or 8 wherein said output wheel is a sliding vane output wheel.

10. The apparatus according to any one of claims 5, 6, 7, or 8 wherein said output wheel is a gear type output wheel.

11. The apparatus according to any one of these claims 5, 6, 7, or 8 wherein said output wheel is a regenerative blower output wheel.

12. The apparatus according to any one of claims 5, 6, 7, or 8 wherein said output wheel is a liquid seal blower output wheel.

13. The apparatus according to any one of claims 5, 6, 7, or 8 wherein said output wheel is a dynamic turbine.

14. The apparatus according to any one of claims 5, 6, 7, or 8 wherein said operating means comprises means for operating said engine in a closed cycle and wherein said exhaust gas feeding means including means for feeding said exhaust gas back to said compressor inlet, and wherein said combustion chamber is an indirect combustion chamber.

15. The apparatus according to claim 1 wherein the sliding vanes of said compressor are made of graphite.

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16. The apparatus according to claim 9 wherein the sliding vanes of said output wheel are graphite.

17. The apparatus according to claim 1 wherein said output wheel includes a rotating member and wherein said operating means includes means for generating a hot-cool operating cycle comprising a hot phase continuously alternating with a cool phase, the gas fed to said output wheel during said hot phase being hot and the gas fed to said output wheel during said cool phase being cooler, said hot phase having a time period less than the time required for said rotating member to soak to substantially the temperature of the hot gas fed to said output wheel during the hot phase.

18. The apparatus according to claim 17 including means for maintaining the speed of said shaft during said cool phase substantially the same as that of said shaft during said hot phase, for providing uniform and continuous output.

19. The apparatus according to claim 17 including means for varying the length of at least one of said hot and cool phases in response to throttle changes.

20. The apparatus according to claim 1 including means for maintaining the temperature of the gas flowing through said output wheel gas inlet higher than the temperature of the gas at any other location in said engine.

21. The apparatus according to claim 1 including means for maintaining the burn termperature in said combustion chamber substantially constant even as the rate fuel is fed to said combustion chamber varies.

22. A method for operating a brayton cycle engine having a combustion chamber and a positive displacement compressor connected by a shaft to an output wheel, said method comprising the steps of feeding a gas to said compressor, feeding compressed gas from said compressor to said combustion chamber, burning fuel in said combustion chamber to heat the gas fed thereto from said compressor,

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feeding hot, compressed gas from said combustion chamber to said output wheel, feeding exhaust gas away from said output wheel, and operating said engine in a substantially constant pressure, Brayton cycle.

23. The method according to claim 22 including operating said engine in an open cycle, including the step of feeding said output gas to ambient.

24. The method according to claim 22 wherein said engine is a closed cycle engine including a closed conduit for continuously circulating a working gas in a closed cycle from said compressor to said output wheel and then back again to said compressor, and wherein said combustion chamber is an indirect combustion chamber, and including the step of indirectly heating said working gas in said combustion chamber.

25. The method according to claim 24 including the step of heat exchanging the exhaust gas from said output wheel with the gas fed from said compressor to said combustion chamber.

26. The method according to claim 22 including the step of compressing said gas in a sliding vane compressor.

27. The method according to claim 22 including compressing said gas in a gear type compressor.

28. The method according to claim 22 including compressing said gas in a regenerative blower.

29. The method according to claim 22 including the step of compressing said gas in a liquid seal blower.

30. The method according to any one of claims 26, 27, 28 or 29 including the step of feeding the gas from said combustion chamber to a sliding vane output wheel.

31. The method according to any one of claims 26, 27, 28, or 29 including the step of feeding the gas from said combustion chamber to a gear type output wheel.

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32. The method according to any one of claims 26, 27, 28 or 29 including the step of feeding the gas from said combustion chamber to a regenerative blower output wheel.

33. The method according to any one of claims 26, 27, 28 or 29 including the step of feeding the gas from said combustion chamber to a liquid seal blower output wheel.

34. The method according to any one of claims 26, 27, 28, or 29 including the step of feeding the gas from said combustion chamber to a dynamic turbine output wheel.

35. The method according to any one of claims 26, 27, 28 or 29 wherein said engine is a closed cycle engine including a closed conduit for continuously circulating a working gas in a closed cycle from said compressor to said output wheel and then back again to said compressor, and wherein said combustion chamber is an indirect combustion chamber, and including the step of indirectly heating said working gas in said combustion chamber.

COMPRESSOR

TURBINE

AIR

14

12

16

OUTPUT SHAFT

EXHAUST GAS

10

BURNER

18

FUEL

FIG. 1

$P_c = 50$ h.p.

$P_T = 150$ h.p.

$\dfrac{P_c}{P_o} = 0.5$

20

24

22

$P_o = 100$ h.p.

FIG. 2A

$P_c = 200$ h.p.

$P_T = 300$ h.p.

$\dfrac{P_c}{P_o} = 2.0$

26

30

28

$P_o = 100$ h.p.

FIG. 2B

FIG. 3

FIG. 4

FIG. 5

NET OUTPUT SHAFT
POWER = 50 h.p.

PRESSURE RATIO = 4.0

INLET (AMBIENT) = 70°F

$\eta_T = 0.9$

$\eta_c = 0.85$

$\gamma = 1.4$

FIG. 6

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FIG. 7

FIG. 8

AIR INLET   EXHAUST GAS

```
        43              42              44        45

   ┌──────────────┐                ┌──────────────┐        48
   │              │                │              │
   │  COMPRESSOR  │────────────────│    POWER     │────────  OUTPUT
   │              │       41       │    WHEEL     │          SHAFT
   └──────────────┘                └──────────────┘

 40
          49    ┌──────────────┐
        ────────│    BURNER    │────────
                │              │  46
                └──────────────┘
                       47
                     FUEL
```

FIG. 9

```
        52                  56              54              60

   ┌──────────────┐                    ┌──────────────┐
   │              │                    │              │
   │  COMPRESSOR  │────────────────────│    POWER     │────────  OUTPUT
   │              │                    │    WHEEL     │          SHAFT
   └──────────────┘                    └──────────────┘

                                   INDIRECT
                HEAT               BURNER
              EXCHANGER   62            74
                                ┌──────────────┐
                    ╱╲╱╲────────│  ~~~~~~~~     │──────
                    ╱╲╱╲        │          68  │
              66              AIR│──→           │
 50                           FUEL│──→          │──→ EXHAUST
                                  │   70        │  72  GAS
                                  └──────────────┘
                                              58
                              64
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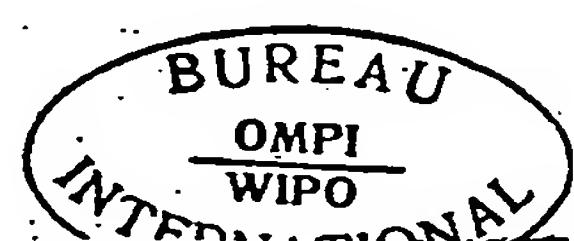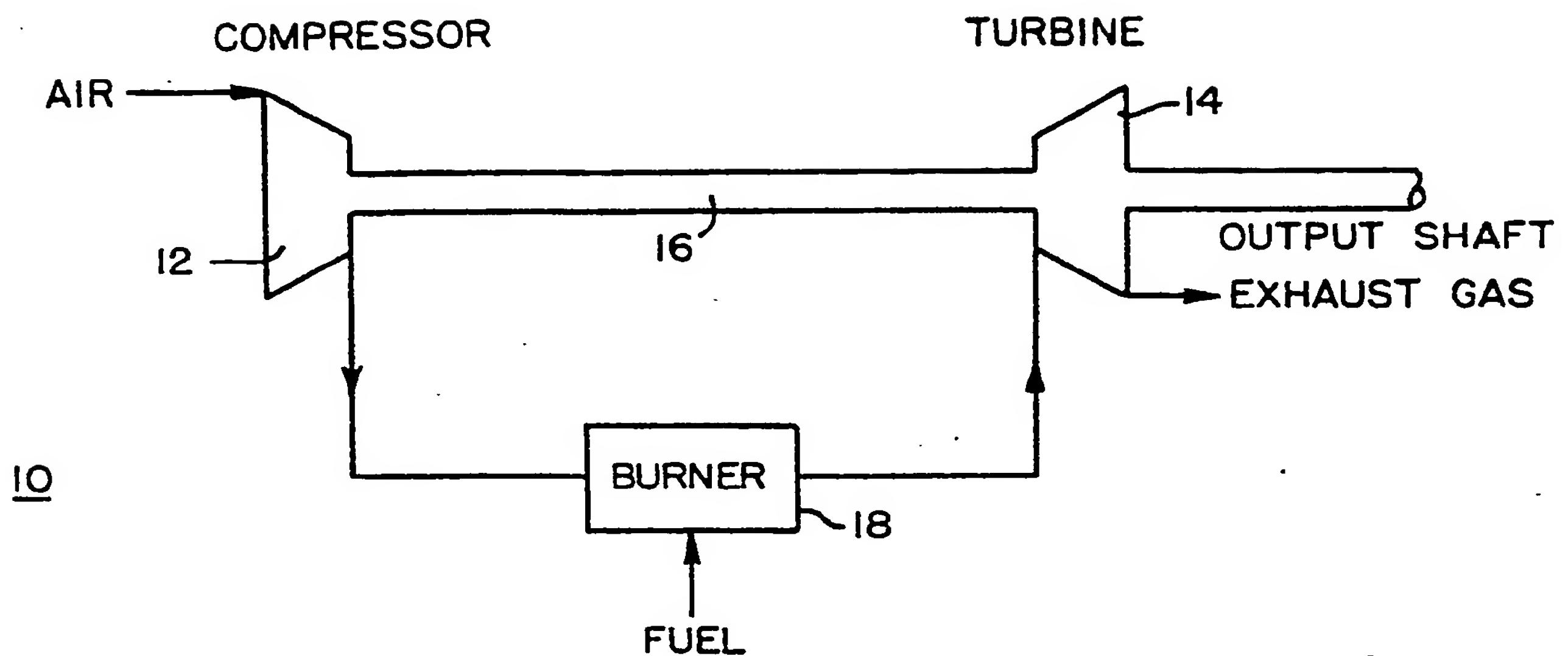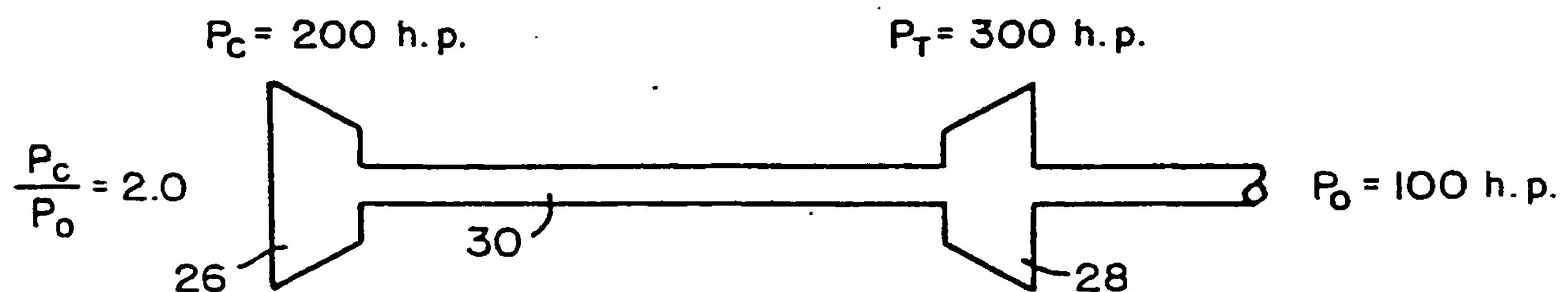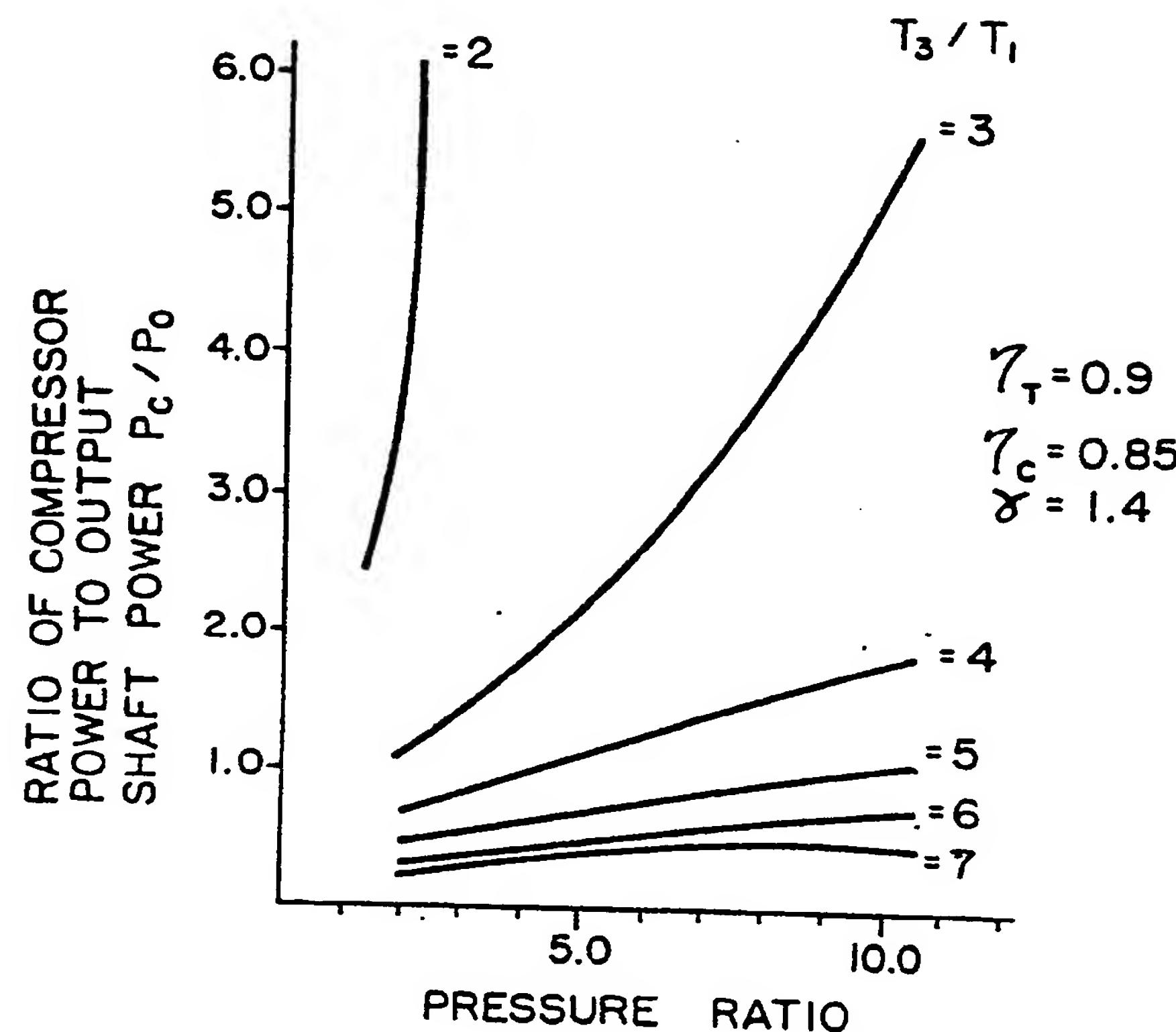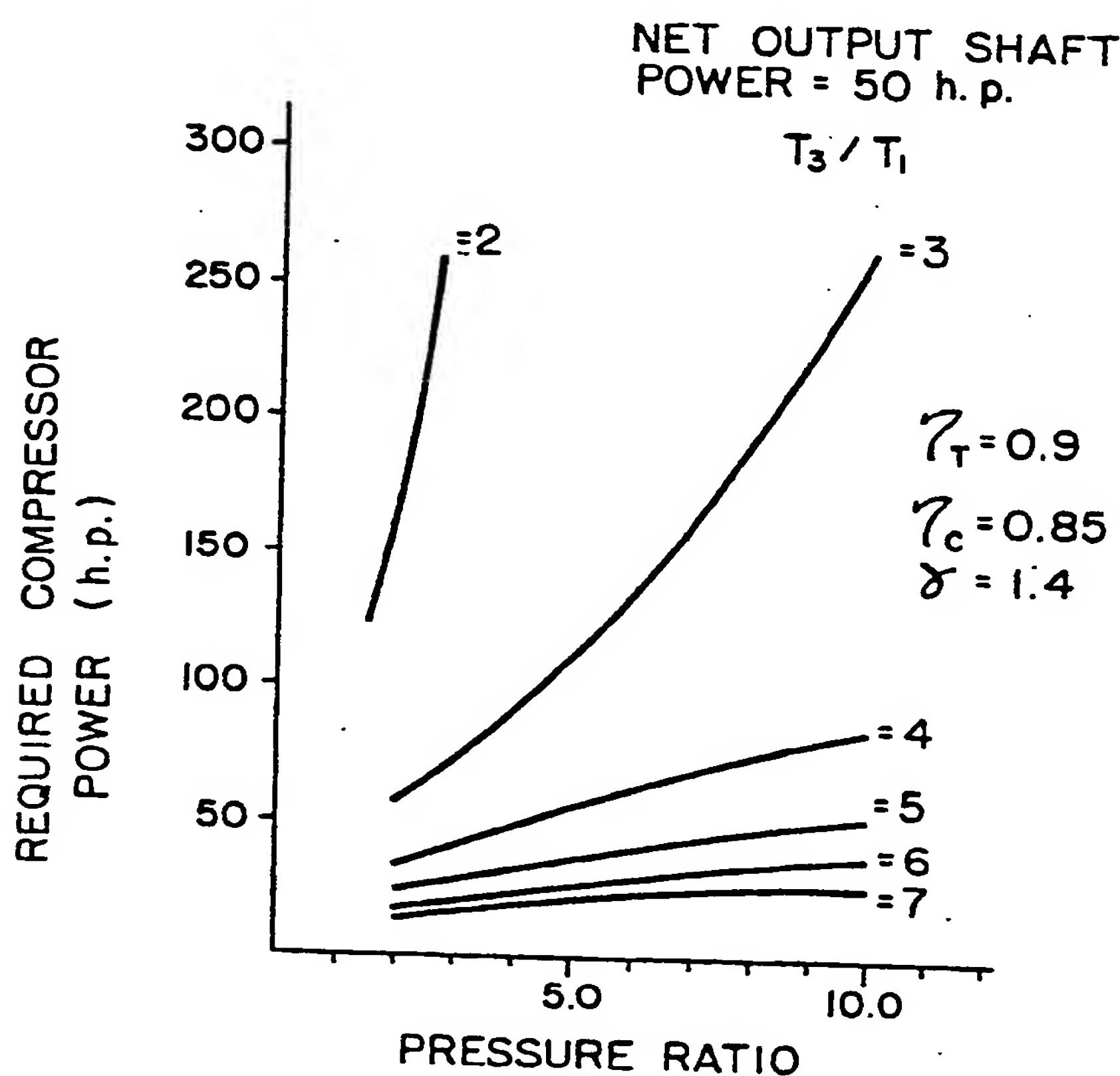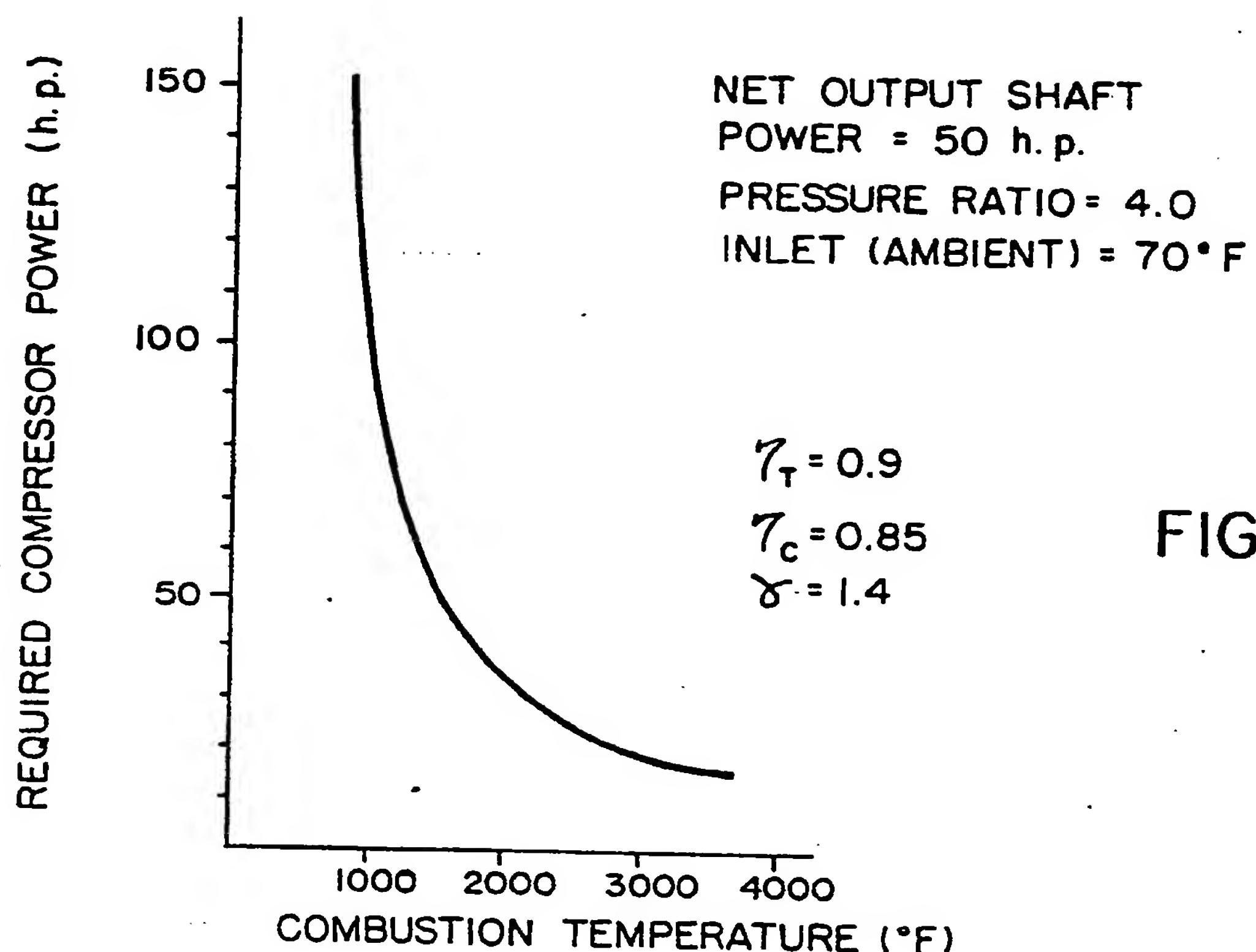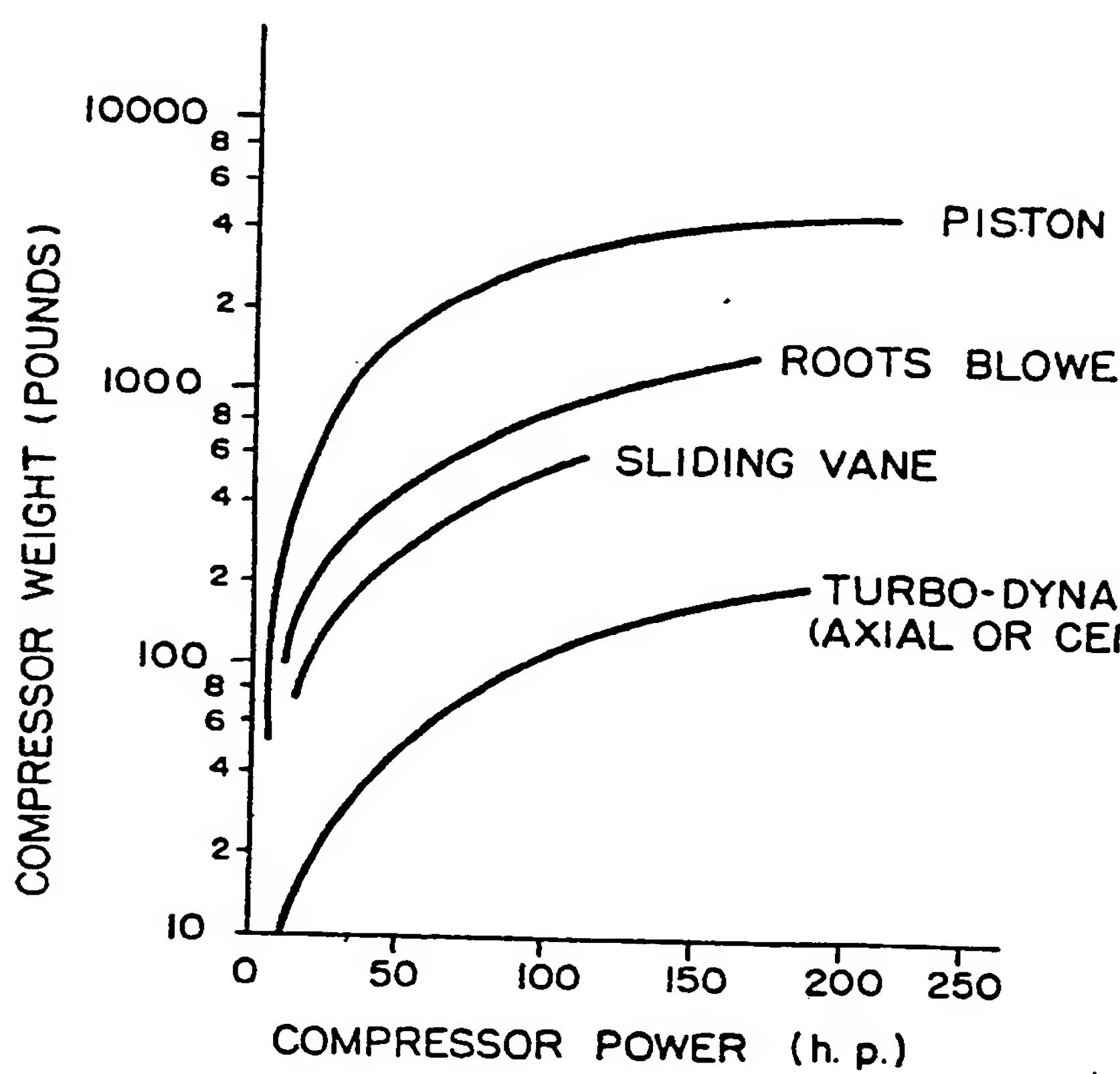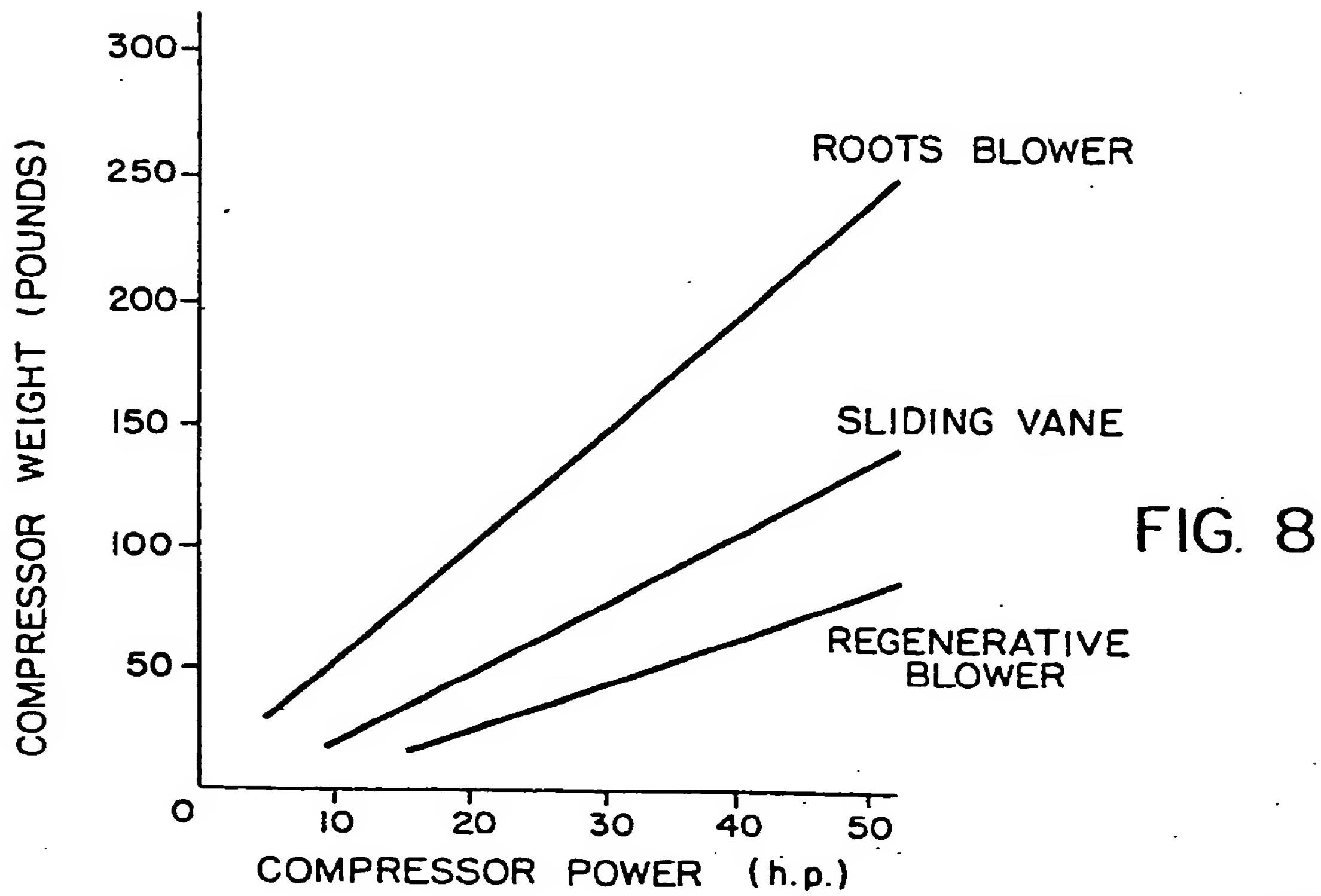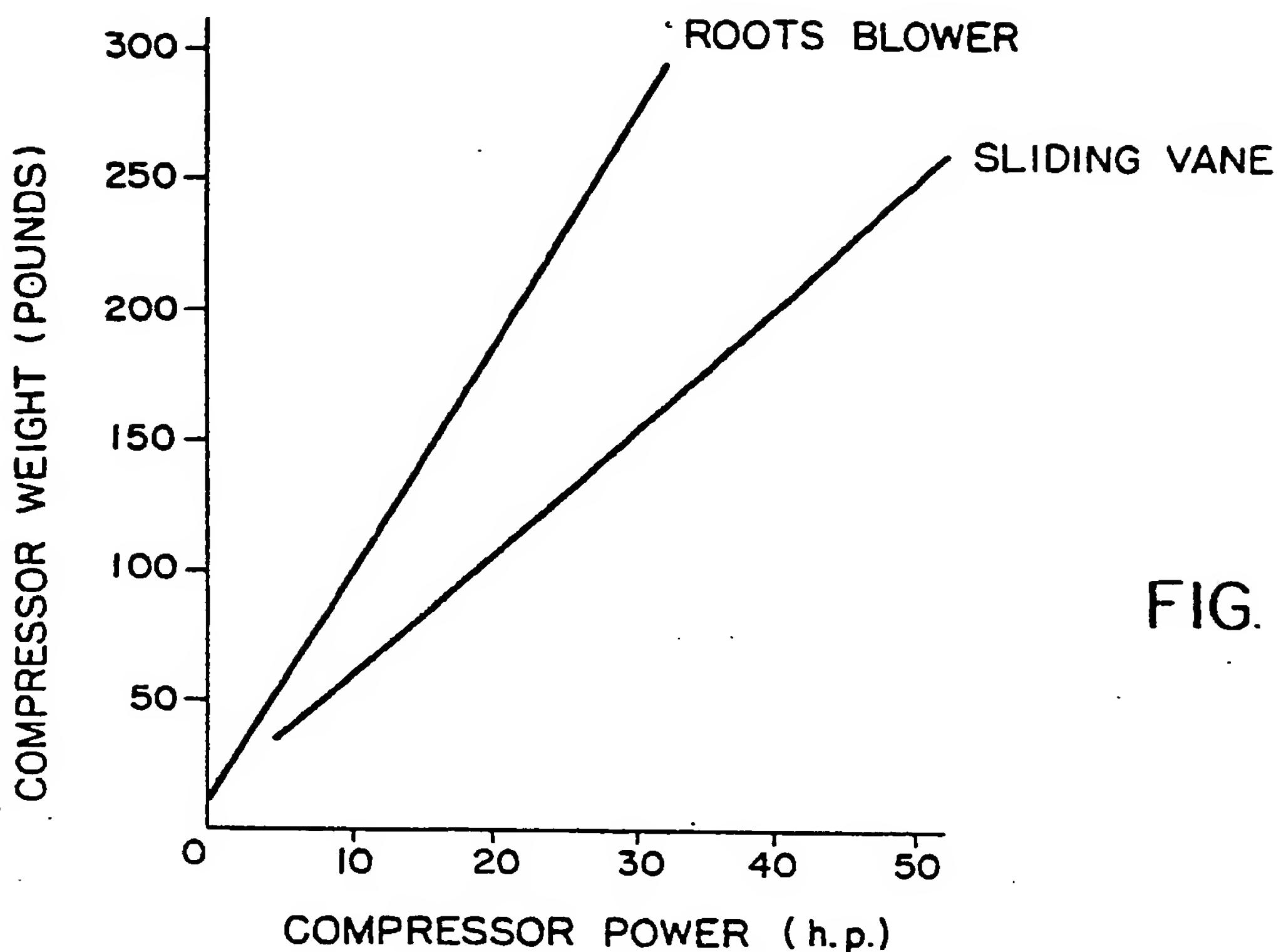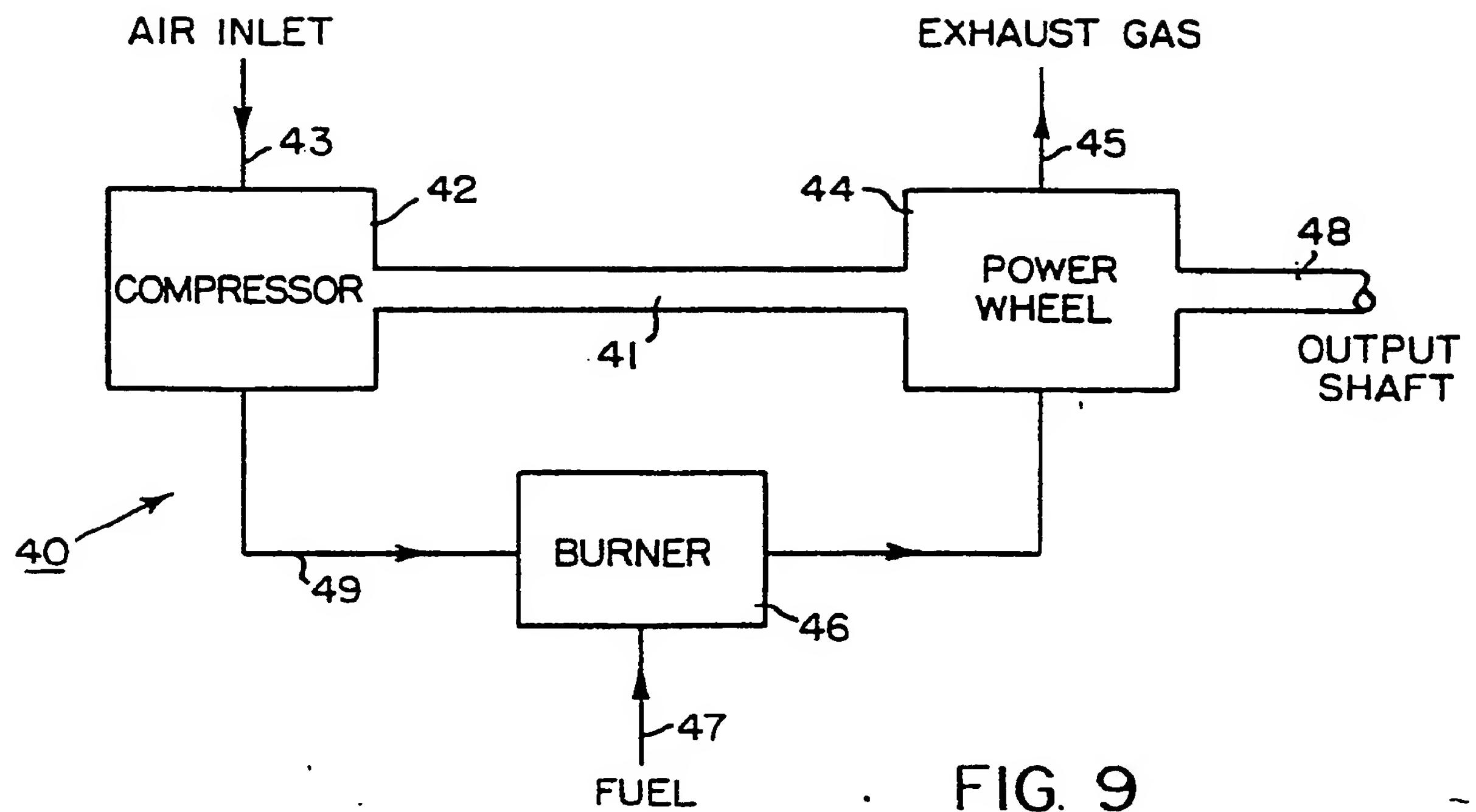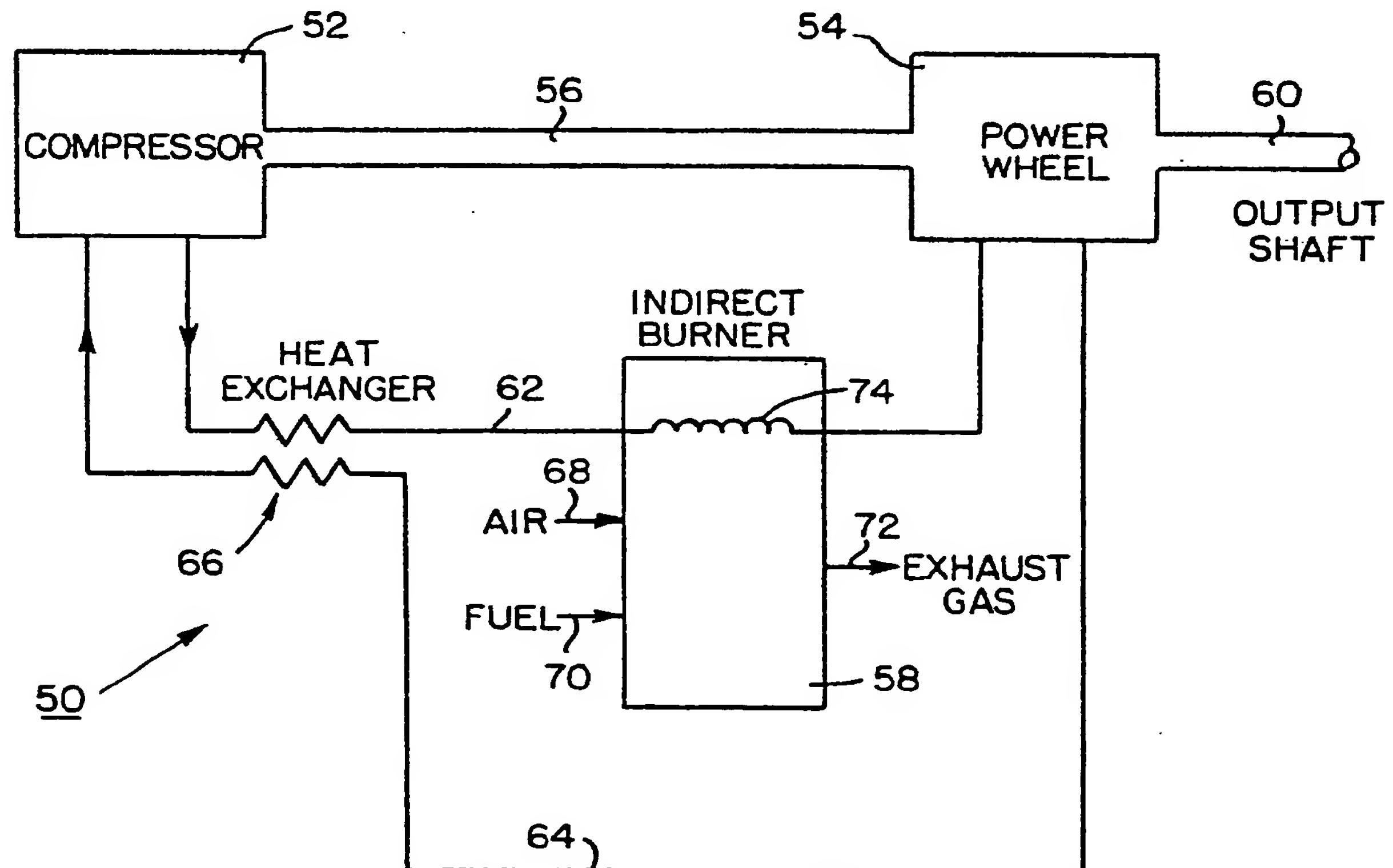
FIG. 10
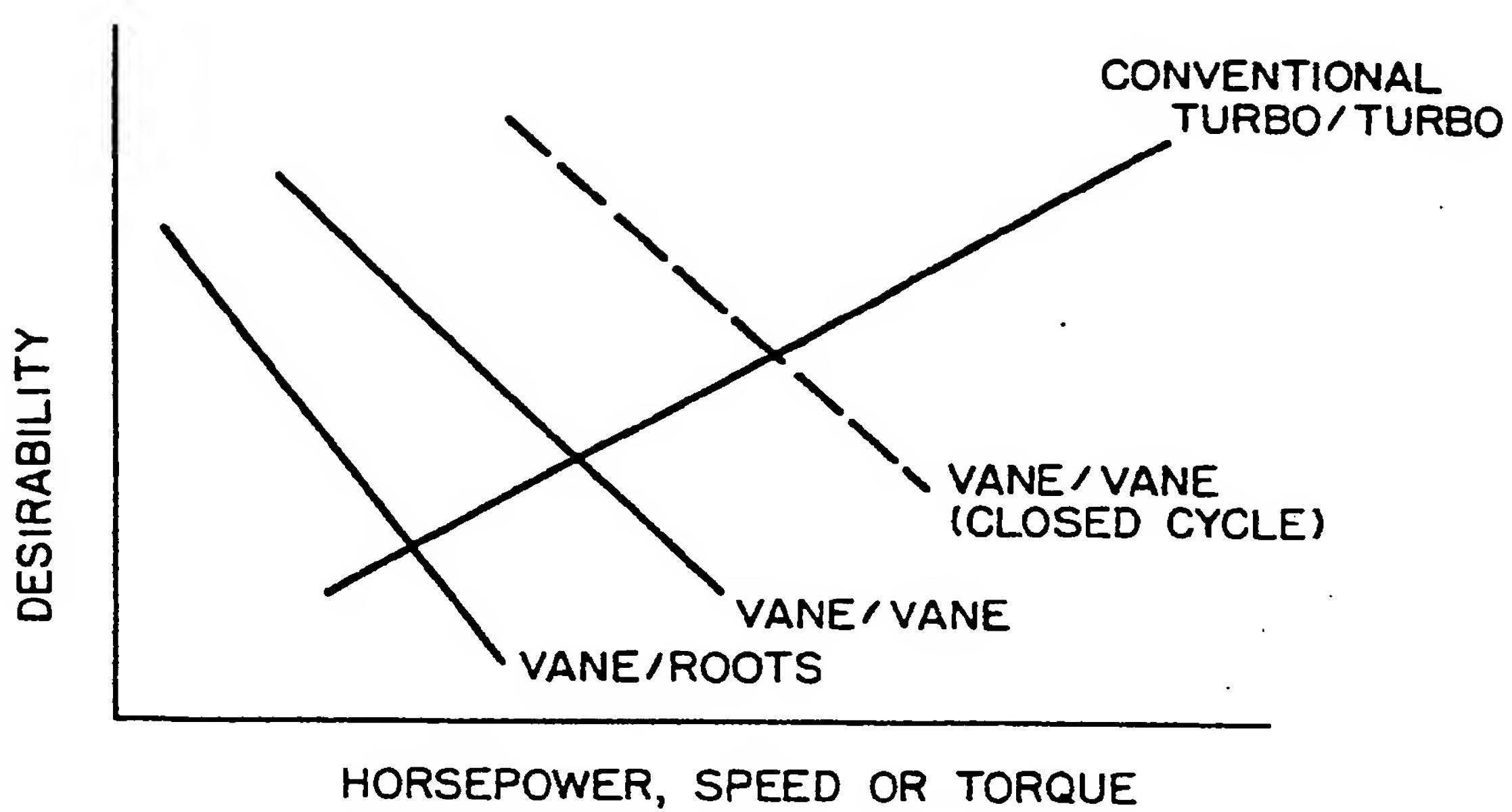
FIG. 11
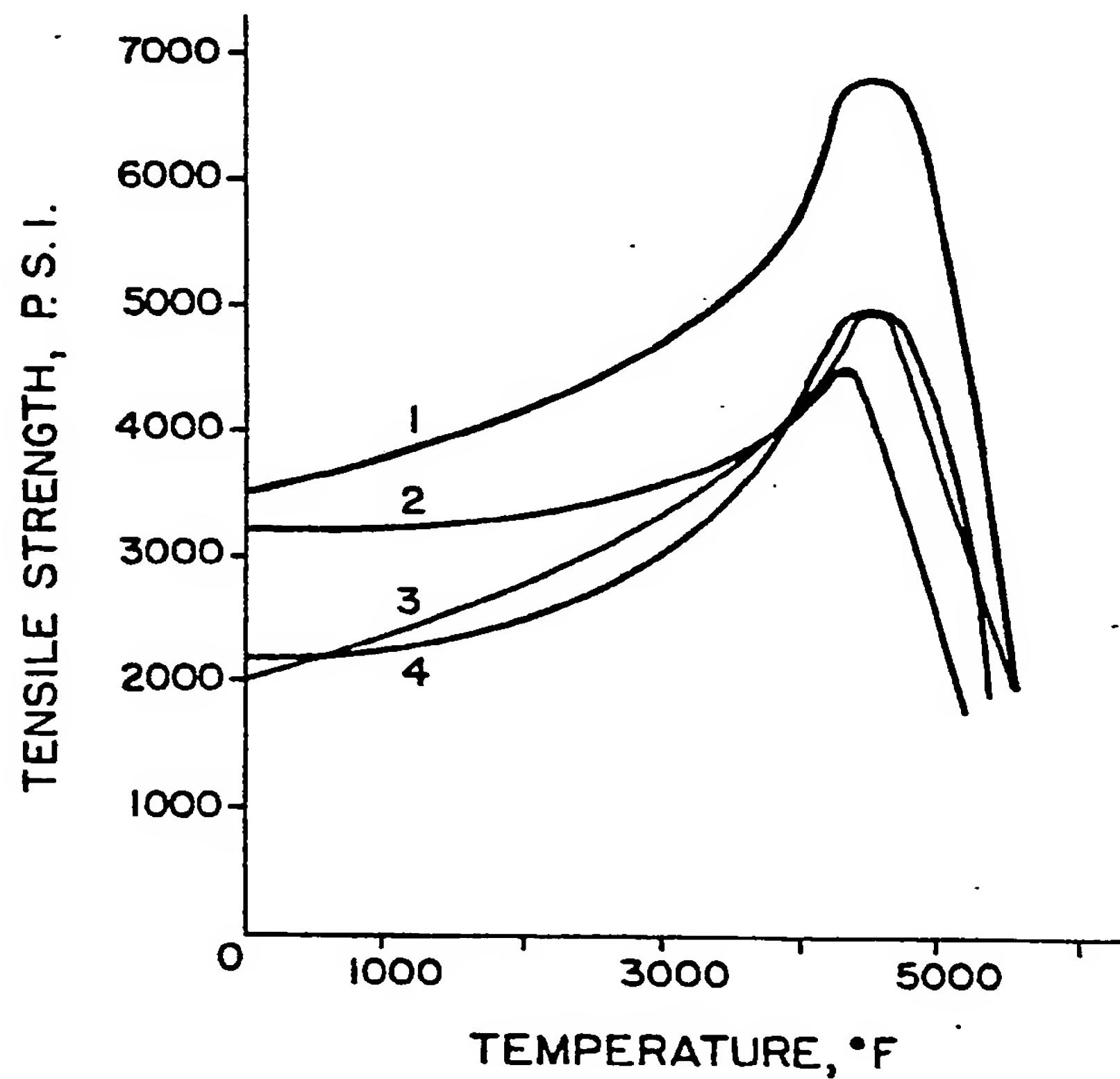
FIG. 12
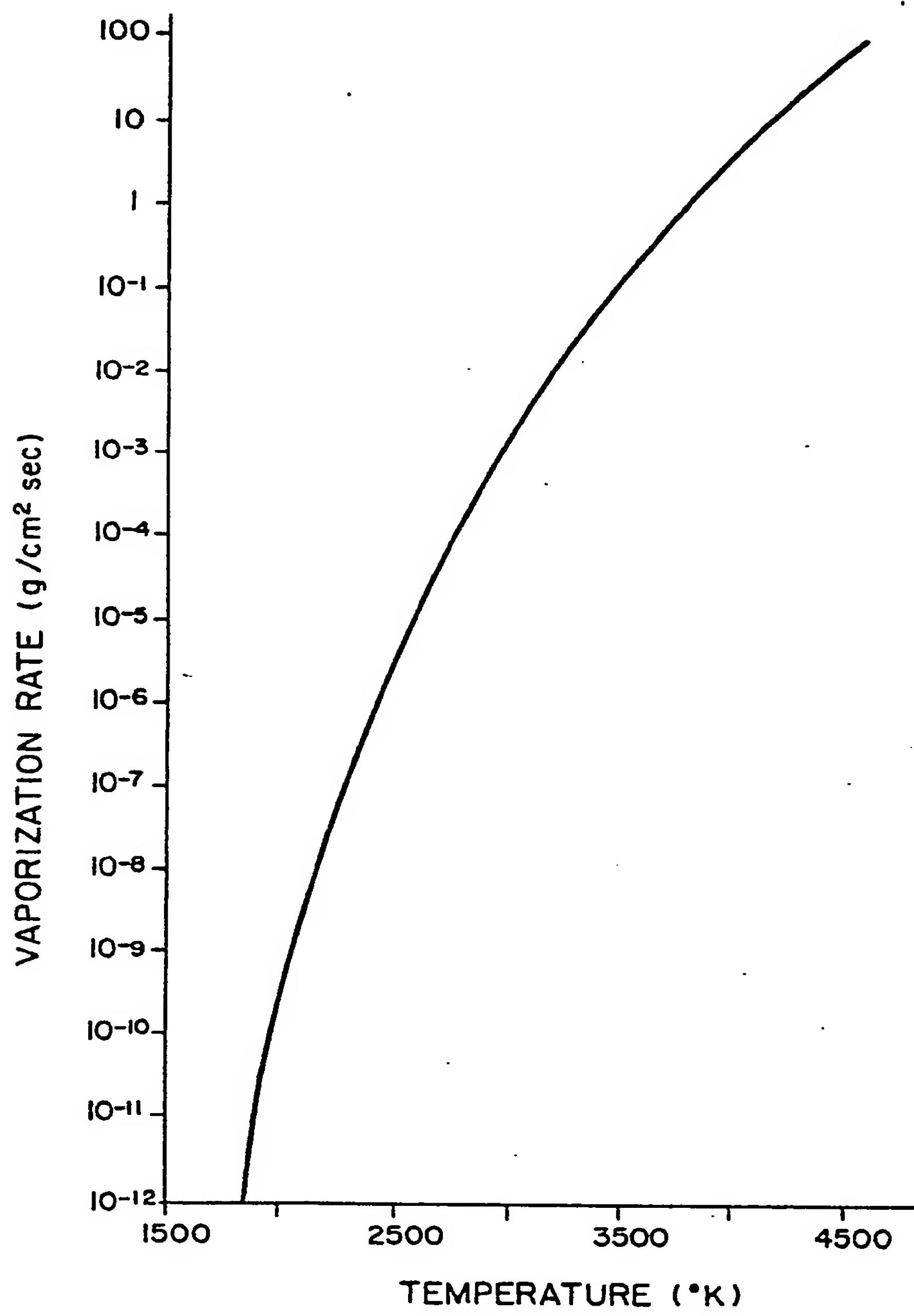
FIG. 13
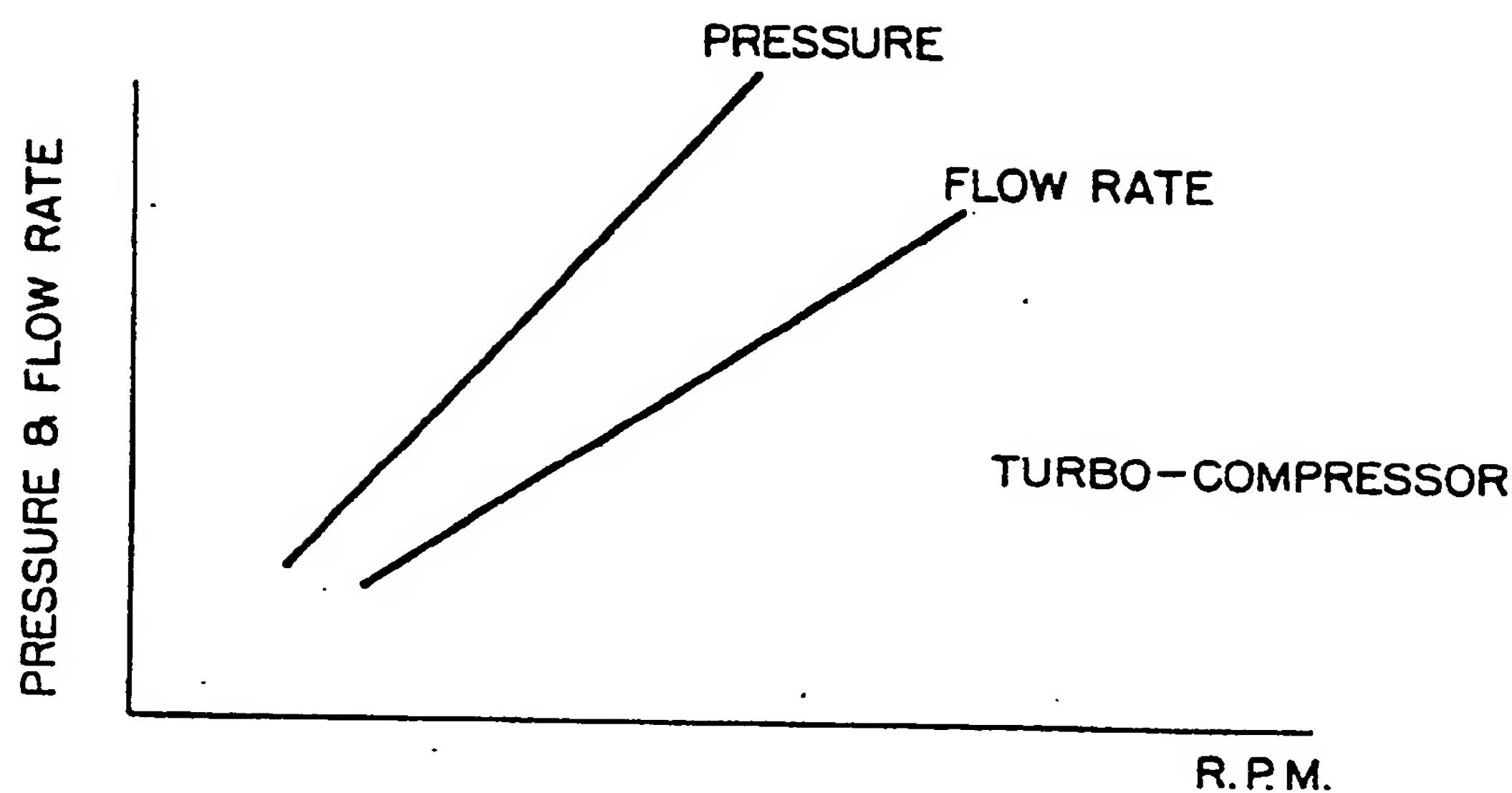
FIG. 14
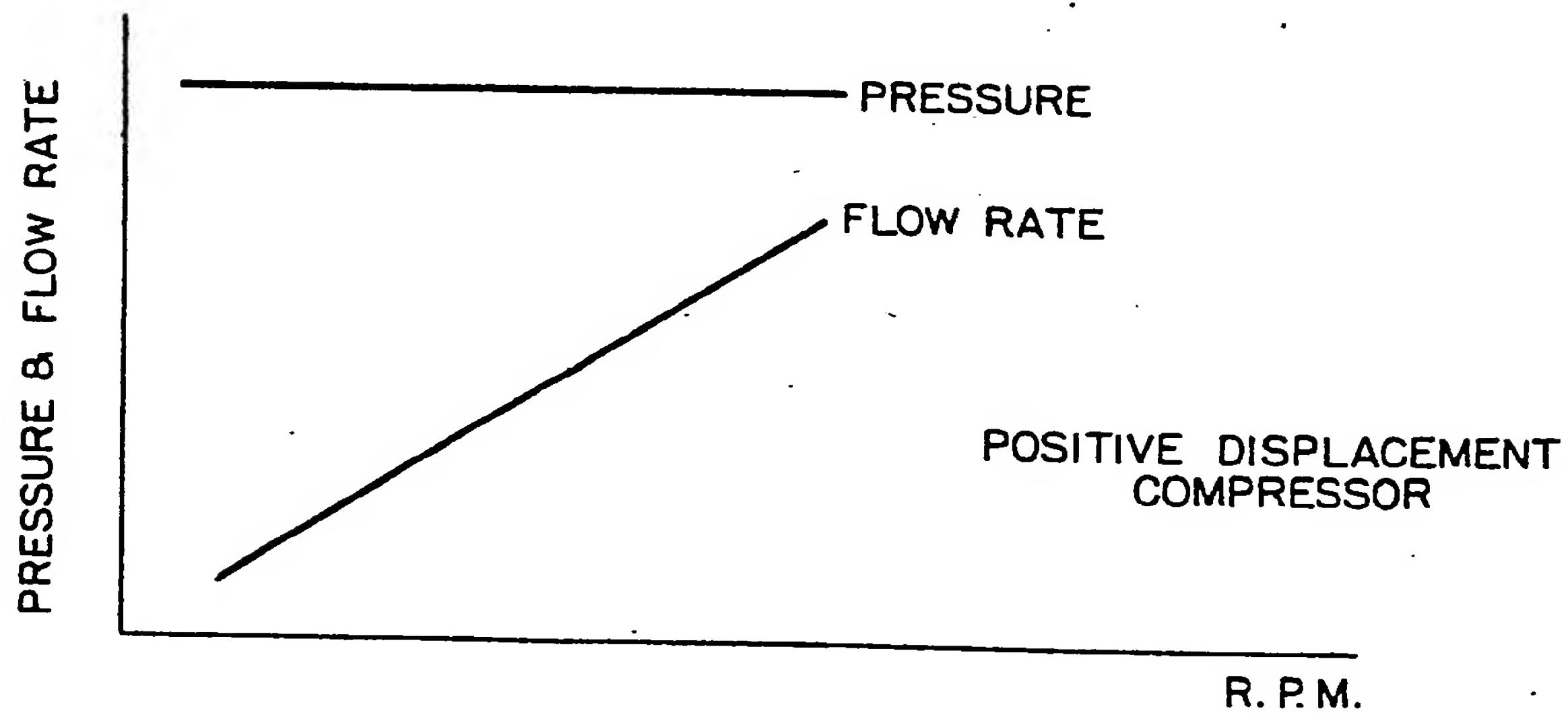
FIG. 15
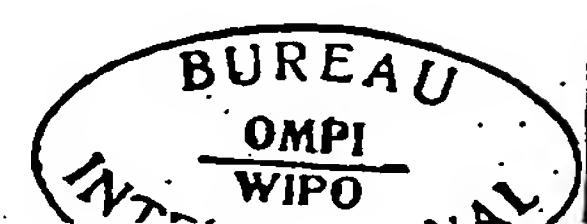
INTERNATIONAL SEARCH REPORT

International Application No **PCT/US79/00324**

I. CLASSIFICATION OF SUBJECT MATTER (If several classification symbols apply, indicate all) ³

According to International Patent Classification (IPC) or to both National Classification and IPC

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Intl. Cl.   F02C 3/04; F02C 5/02
U.S. Cl.   60/39.02; 60/39.45; 60/39.63; 60/39.76
```

II. FIELDS SEARCHED

Minimum Documentation Searched ⁴

Classification System	Classification Symbols
U.S.	60/39.02, 39.45, 39.6, 39.63, 39.06, 39.76 123/202, 204

Documentation Searched other than Minimum Documentation
to the Extent that such Documents are Included in the Fields Searched ⁵

III. DOCUMENTS CONSIDERED TO BE RELEVANT ¹⁴

Category *	Citation of Document, ¹⁶ with indication, where appropriate, of the relevant passages ¹⁷	Relevant to Claim No. ¹⁸
X	US,A, 3,584,459, Published 15 June 1971, Amann	1,2,4-13, 15,16,20-23, 26-32, 34
X	US,A, 3,057,157, Published 9 October 1962, Close	5,6,8-10, 12,26,27, 29-31,33
X	US,A, 3,191,852, Published 29 June 1965, Kaatz et al	15,16
X	US,A, 3,501,913, Published 24 March 1970, Brille	1,17-19
X	US,A, 3,672,164, Published 27 June 1972, Pieper	1,17-19
A	US,A, 2,461,757, Published 15 February 1949, Moores	1,2,4-13, 15-23,26-34
A	US,A, 3,765,171, Published 16 October 1973, Hagen et al	1,2,4-13, 15-23,26-34
A	US,A, 3,558,236, Published 26 January 1971, Bylsma	7,9-14,16, 28,30-35

* Special categories of cited documents: ¹⁵

"A" document defining the general state of the art

"E" earlier document but published on or after the International filing date

"L" document cited for special reason other than those referred to in the other categories

"O" document referring to an oral disclosure, use, exhibition or other means

"P" document published prior to the International filing date but on or after the priority date claimed

"T" later document published on or after the International filing date or priority date and not in conflict with the application, but cited to understand the principle or theory underlying the invention

"X" document of particular relevance

IV. CERTIFICATION

Date of the Actual Completion of the International Search ²	Date of Mailing of this International Search Report ²
17 August 1979	1 9 SEP 1979

International Searching Authority ¹	Signature of Authorized Officer ¹⁰
ISA/US	Carlton E. Croyle

Form PCT/ISA/210 (second sheet) (October 1977)

FURTHER INFORMATION CONTINUED FROM THE SECOND SHEET		
A	US,A, 3,973,865, Published 10 August 1976, Mugele	7,9-14,16, 28,30-35
X	GB,A, 947,822, Published 29 January 1964, Birmann	1,17-19
A	AT,B, 140,501, Published 11 February 1935, Pistl	1,2,4-13, 15-23,26-34

V.☐ OBSERVATIONS WHERE CERTAIN CLAIMS WERE FOUND UNSEARCHABLE [10]

This international search report has not been established in respect of certain claims under Article 17(2) (a) for the following reasons:

1.☐ Claim numbers _____, because they relate to subject matter [12] not required to be searched by this Authority, namely:

2.☐ Claim numbers _____, because they relate to parts of the international application that do not comply with the prescribed requirements to such an extent that no meaningful international search can be carried out [13], specifically:

VI.☒ OBSERVATIONS WHERE UNITY OF INVENTION IS LACKING [11]

This International Searching Authority found multiple inventions in this international application as follows:

I. Brayton cycle engine operating in open cycle.
II. Brayton cycle engine operating in closed cycle.

1.☐ As all required additional search fees were timely paid by the applicant, this international search report covers all searchable claims of the international application.

2.☐ As only some of the required additional search fees were timely paid by the applicant, this international search report covers only those claims of the international application for which fees were paid, specifically claims:

3.☒ No required additional search fees were timely paid by the applicant. Consequently, this international search report is restricted to the invention first mentioned in the claims; it is covered by claim numbers: 1,2,4-13,15-23,26-34

Remark on Protest

☐ The additional search fees were accompanied by applicant's protest.

☐ No protest accompanied the payment of additional search fees.

Form PCT/ISA/210 (supplemental sheet (2)) (October 1977)